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Using supercritical heat recovery process in Stirling engines for high thermal efficiency

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Abstract

Stirling engine, using a composite working fluid, such as two-component fluid: gaseous carrier and phase-change component and single multi-phase fluid as the working fluid is studied to get high thermal efficiency. In Stirling engine with a composite fluid, a thermodynamic supercritical heat recovery and heating process is proposed and demonstrated to improve the heat transfer of the heat regenerator and cooler of common gaseous Stirling engine. The criteria for the choice of the working fluids are also formulated. © 2001 Elsevier Science Ltd. All rights reserved.

Keywords: Stirling engine; Supercritical power cycle; Composite working fluid; Regenerative process

1. Introduction

The theoretical thermal efficiency of Stirling engines with the regenerative process, invented early in 1816 [1], is equal to the Carnot efficiency. The actual thermal efficiency of gaseous Stirling engines is low (usually lower than 0.4) even though with a very high average charge pressure (usually 15.0 MPa) and operated at a high heat source temperature [2–4]. However, the freedom to choose the most appropriate working fluids and the opportunity to incorporate a sophisticated combustor with exhaust heat recuperation means that the Stirling engine can approach its

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theoretical efficiency more closely than many other engines. At least in the medium- and low-power applications for which the large and complex system of a central steam power plant is not practical.

So far, either gas [2–5] or liquid [6] has been used as the working fluid in Stirling engines. Composite working fluids, such as two-component working fluids: the gaseous carrier and the phase-change component, chemically reactive working fluids and single multi-phase working fluids were proposed by Walker [3] and studied [7–9] for the possibility of achieving a high specific power at a moderate mean pressure level, improved heat transfer as well as higher thermal efficiency.

However, it was pointed out by West [4] that although the specific power of Stirling engines may be significantly improved by the use of compound working fluids, the test with the apparatus using air and water was most disappointing for no way had been found to maintain a uniform distribution of the two components in the cylinder. A conventional regenerator would also not work with a condensing fluid because the saturation temperature is, unfortunately, lower during the low pressure of the expansion part of the cycle (when heat is normally stored in the regenerator) than during the high pressure of the compression phase, when the heat stored in the regenerator should be returned to the working fluid. So the heat regeneration process should be improved when a composite working fluid is used as the working fluid.

Here we show, using a supercritical heat-recovery and heating process, a composite working fluid can be effectively used in Stirling engine, especially retaining the intrinsic high thermal efficiency of the cycle. We will also discuss the changes of the proposed system from the common Stirling engine.

2. Heating of supercritical fluid

The thermodynamic critical state is the most important characteristic of a real working fluid (gas or liquid) [10]. With a supercritical pressure, the continuous phase change from liquid to vapor (gas) with no evaporation will take place if the temperature of the liquid rises up to some extent when heated [11–13].

Considering constant supercritical pressure, the specific volume of the liquid (from point k to point l), illustrated by Fig. 1, will expand with the increase of the temperature. The continuous phase change from liquid to vapor (gas) with no evaporation at point l takes place if the temperature of the liquid rises to the temperature of point l when heated. The vapor (gas) will then reach the superheated state (point m) when its temperature is raised higher than the temperature at point l.

Furthermore, considering a pipe with a temperature gradient in length and with a capacity of heating, the continuous distribution from liquid to gas can be visualized if the pressure in the pipe is at the supercritical pressure. With continuous heat addition to the supercritical fluid, the volume of liquid phase reduces while the volume of gas phase increases, and therefore, the pressure in the pipe and the temperature of gas phase increases. This feature of supercritical fluid is used to improve the heat regeneration process, or heat-recovery process, of Stirling engines.

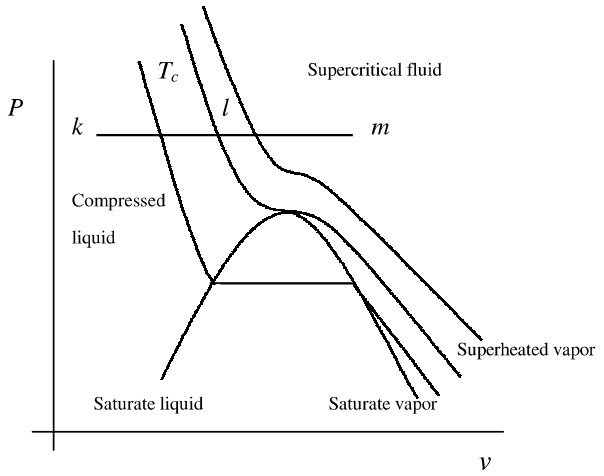

Fig. 1. Heating of supercritical fluid.

3. Stirling engine with two-phase two-component working fluid

Compared to the configuration of common gaseous Stirling engines [2–4,14–17], an improved configuration of Stirling engines, when two-phase two-component working fluids adopted [7–9], was proposed by adding a by-pass supercritical heat-recovery process and replacing the conventional regenerator by a heat exchanger shown in Fig. 2. The condensed liquid, lifted up to the supercritical pressure by the feed pump, enters into the shell-side of the heat exchanger where

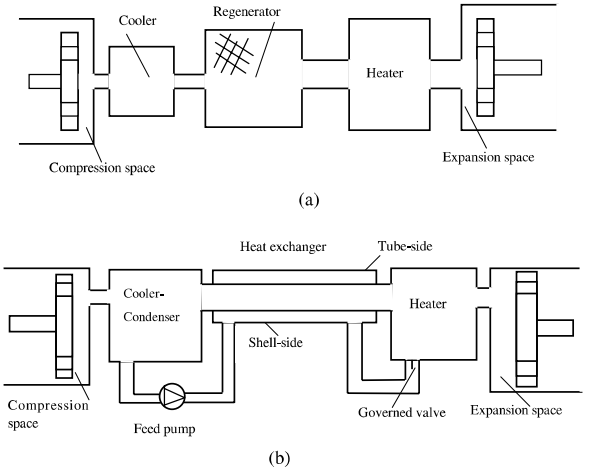

Fig. 2. Configuration of common Stirling engines and its improvement with two-phase two-component working fluid. (a) Configuration of common gaseous Stirling engines. (b) Configuration of Stirling engines with two-phases two-component working fluid.

the continuous phase-change from liquid to gas takes place when heat supplied from the flow of the mixture: the gaseous carrier and superheated vapor of the phase-change component in the tube-side. The function of the heat-recovery process in the heat exchanger is the same as that of the regeneration process in the regenerator of common gaseous Stirling cycles.

It is evident that the temperature gradient in the length of the heat exchanger exists when the counterflow streams between the tube-side and shell-side move through the heat exchanger respectively. The pressure of the supercritical fluid in the shell-side of the heat exchanger rises because the governed is closed before the expansion part of the cycle. When the expansion part of the cycle begins, the governed valve is open to the heater and the superheated gas in shell-side of heat exchanger flows into heater, mixing with the gaseous carrier or inert component, and is heated again and then the mixture comes into the expansion space to produce work. The governed valve will be closed to keep the pressure in the shell-side of the heat exchanger over the critical pressure at all times.

The two-phase, two-component working fluid was assumed to consist of two components, one with very low critical temperature, such as hydrogen and helium, behaving at all times as a perfect gas, and the other existing in the liquid state at low temperatures and in vapor state at high temperatures where it was further assumed to behave as a perfect gas. The specified mass ratio $\beta = m_V/m_a$ is defined as the ratio of the phase-change mass from liquid to vapor to the remainder in the system [3].

Larger mass ratio β means larger pressure drop of the system, and consequently greater work output. The mass ratio β depends on the properties, especially the critical state parameters of the phase-change component.

4. Stirling engine with single multi-phase fluid

Removal of the inert gaseous carrier changes the configuration, as shown in Fig. 3. The compression space is omitted and two governed valves, intake valve and exhaust valve, are needed [14,16]. The T–S diagram of the cycle with single multi-phase working fluids is different from that of the idealized Stirling cycle [2–4]. Two cases of the T–S diagrams are demonstrated in Fig. 3. The main difference between the two cases is the process 3′–4, which produces work with pressure drop while the temperature remains constant because of the consecutive heat addition in case 1. Thus, the net available work in case 1 is greater than that in case 2 for given limits of pressure, volume, and temperature. However, most heat sources for conversion into mechanical and electrical energy are sources of a sensible heat, i.e., of variable temperature and therefore the wholly gliding temperature in T–S diagram of case 2 is very appropriate for a sensible heat source with a great effective temperature difference for heat transfer. The case 2 is confined in the following discussion. The cycle is composed, therefore, of the following processes.

Process 4–1: adiabatic expansion in expansion space for work output;
Process 1–1′ and 2–3: heat recovery in heat exchanger;
Process 1′–2: heat rejection to the external dump in condenser with the condensing process;
Process 2–2′: pressure lift of the condensed liquid by feed pump;
Process 3–4: heat addition in heater from the external heat source.

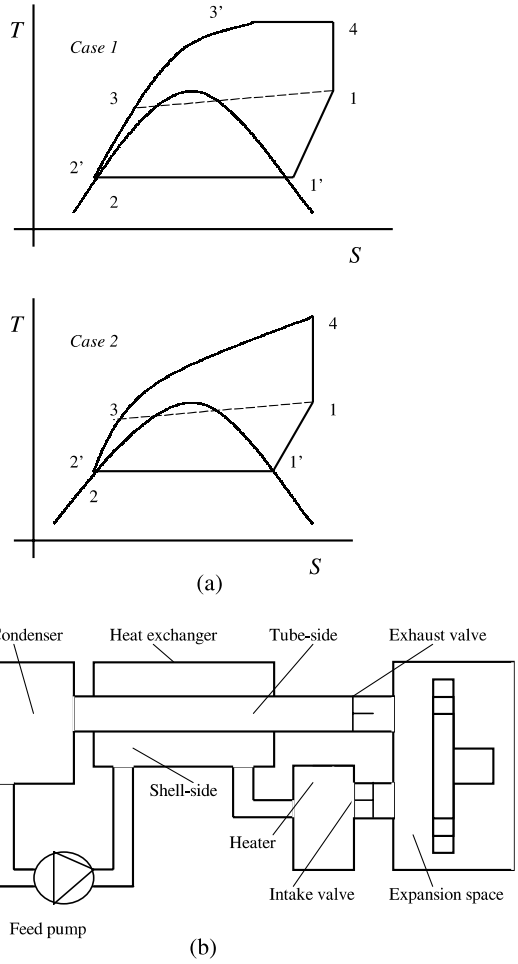

Fig. 3. Configuration of Stirling engine with single multi-phase working fluid and its T–S diagrams. (a) Two cases of T–S diagrams. Case 1: the volume of condenser is an order of magnitude larger than that of expansion space and the total gas-phase volume between heat exchanger and heater is an order of expansion space; case 2: the volume of condenser is an order of magnitude larger than that of expansion space and the total gas-phase volume between heat exchanger and heater is also an order of magnitude larger than that of expansion space. (b) Configuration of Stirling engines with single multi-phase fluids.

After the sensible heat of the working fluid (superheated vapor) swept from the expansion space is recovered in the heat exchanger between sub-process 1–1′ and 2′–3 (the pressure at point 2′ is different from the pressure at point 2 while the temperatures at two points respectively are the same), the latent heat of the working fluid, with phase change from gas (vapor) to liquid, is transferred at T_2 to the external dump in the condenser in process 1′–2.

Considering 1 kg working fluid in the cycle in Fig. 2(b), the work output in the adiabatic process 4–1 is written as

$$w_{\mathrm{ad}} = \frac{\kappa}{\kappa - 1} RT_4 \left(1 - \frac{1}{\varepsilon_T}\right) = \frac{\kappa}{\kappa - 1} RT_4 (1 - \varepsilon_p^{((1-\kappa)/\kappa)}) \tag{1}$$

where k is the isentropic exponent; R is the characteristic gas constant, kJ/(kg K); T_4 is the heat source temperature, K; ε_T is the temperature ratio T_4/T_1; ε_p is the pressure ratio p_4/p_1, $\varepsilon_T = \varepsilon_p^{(\kappa-1/\kappa)}$. The adiabatic efficiency, η_{ad}, is used for the calculation of the reduction of work output because of the irreversibility in process 4–1. Thus the net work from the cycle is

$$w = w_{\mathrm{ad}} \eta_{\mathrm{ad}} - w_{\mathrm{P}} \tag{2}$$

where w_{P} is the actual work input for feed pump. The heat rejection from the cycle, q_{r}, is equal to the latent heat of the working fluid, r, kJ/kg, because the heat rejection from the cycle only happens in condensing process 1′–2.

$$q_{\mathrm{r}} = r \tag{3}$$

So the thermal efficiency of the cycle is

$$\eta = \frac{w}{w_{\mathrm{ad}} \eta_{\mathrm{ad}} + r} \tag{4}$$

Rewriting Eq. (4) as

$$r = \frac{w}{\eta} - w_{\mathrm{ad}} \eta_{\mathrm{ad}} \tag{5}$$

In the phase change regime, a given saturate pressure or temperature is a function of the latent heat of the working fluid in the T–S diagram.

$$p_1 = p_2 = p(r), \qquad T_2 = T(r) \tag{6}$$

Thus, the minimum pressure, p_2, one of the important cyclic parameters of Stirling engines should be decided with no freedom by the cyclic parameters in Eqs. (4), (2) and (1).

It should be mentioned that the calculation of the latent heat of the working fluid and the thermal efficiency of the cycle is first based on the calculation of the cyclic parameters. So a trial and error calculation process is needed for the choice of cyclic parameters of Stirling engines, start value of η using the Carnot efficiency. This process can be programmed. The thermal efficiency then could approach the theoretical value if the latent heat of the working fluid approaches the value calculated.

In summary, a suitable working fluid, its appropriate condensing temperature (pressure), and consequently the low latent heat, can be chosen for the cycle to promote the thermal efficiency of Stirling engine to a high value. The minimum pressure, p_2, one of the important cyclic parameters of Stirling engines should be decided with no freedom.

For example, sulfur hexafluoride (SF_6) is utilized as the working fluid. The isentropical exponent is $\kappa = 1.20$, which is the average value at the range of interest [10]. The adiabatic efficiency in expansion process and feed pump efficiency are assumed 0.82 and 0.72, respectively, which are the efficiencies of general turbines and feed pumps [18,19]. The optimized condensing temperature, 308 K (or the condensing pressure, 2.97 MPa) is determined at high temperature, 600°C, which is under the chemically stable temperature [20] and the calculated thermal efficiencies are 0.415, 0.421, 0.424 for given pressure ratios in the expansion process, 4.0, 4.5, 5.0, respectively.

It is worth to be mentioned that the optimum condensing temperature is about 10 K lower than the critical temperature of sulfur hexafluoride, 318.73 K [10] and this temperature difference between the optimum condensing temperature of the cycle and the critical temperature of the phase-change component is simultaneously verified by other working fluids, such as propane, C_4F_{10}, and R125, according to our preliminary calculation. This also means that the optimum condensing pressure of the cycle is near the critical pressure of phase-change component.

5. Criteria for the choice of phase-change component

According to our preliminary calculation, the critical temperature of phase-change component is about 10 K higher than the optimum condensing temperature of a suitable phase-change component and therefore the critical temperature of phase-change component is expected under a regime: higher than the ambient temperature so that the phase-change component can be condensed at low temperature, but not too high, for example up to 100°C, to reduce the exergy loss of the condenser because the exergy loss of the condenser is proportional to the temperature difference between the condensing temperature of the phase-change component and the mean temperature of the coolant. The optimum latent of the phase-change component at the minimum pressure is determined by Eq. (5) because of the heat rejection of the cycle only happening in the condensing process.

The optimized condensing pressure of phase-change component is also near the critical pressure, as discussed above. So at a constant pressure ration of the outlet pressure to the inlet pressure of feed pump, lower the critical pressure of phase-change component and consequently lower the condensing pressure, lower pressure difference between the outlet and inlet of feed pump and as a result lower work input for feed pump and consequently higher thermal efficiency of the cycle.

Thirdly, the phase-change component should be friendly with environment and safe.

So water, which had always been used for phase-change component, is not a good fluid due to its very high critical pressure and temperature. Some potential fluid, such as ethane, propane and sulfur hexafluoride referred to in Ref. [10] and HFC refrigerants [21] may be candidates of the phase-change components.

6. Discussions

The cooler, heat regenerator and compression space in common gaseous Stirling engine are replaced by the condenser, recovery heat exchanger and feed pump, respectively, in the new proposed system using single multi-phase fluid as the working fluid. The condensing process occurring to the phase-change component during the cycle is associated with high rate of heat transfer and consequently, the process of compression in the feed pump is likely to approximate to isothermal condition more closely than in the gaseous machine. The heat exchanger for heat recovery process, similar to heat regeneration, is a shell-tube type heat exchanger and thus the design process for heat recovery is simplified.

Even if the high temperature, 600°C, in our preliminary calculation is lower than the heating temperature of gaseous Stirling engine, usually 750°C, the calculated thermal efficiency is still encouraging, higher than 0.4. If the high temperature is 750°C, the thermal efficiency is 0.458 for the given pressure ration in the expansion process, 4.0 and the condensing temperature, 308 K, assumed the adiabatic efficiency in expansion process and pump efficiency of the feed pump, 0.82 and 0.72, respectively.

However, chemical stabilities of working fluids at high temperatures should be taken into consideration. Sulfur hexafluoride is chemically stable up to 600°C and will be dissociated when temperature is higher than 650°C [20]. Although chemically reactive working fluid, for example, nitrogen tetroxide, was studied for the working fluid of gaseous Stirling engine by Walker [3] and the dissociation reaction of sulfur hexafluoride is also the reaction characteristic of increase in the number of moles, we would rather confine the high temperature under the chemically stable temperature of fluid in preliminary study before further study of the characteristic of the dissociation reaction is carried out.

The pressure ratio in the expansion process slightly influences the calculated thermal efficiency of the cycle. The pressure ratios in the expansion process for our preliminary calculation approach the pressure ratios of the outlet pressure to the inlet pressure of microturbines [22,23]. So the action of expansion space in the new proposed system may be replaced by a microturbine. In addition to the similar pressure ratio, further study should be carried out when microturbine is used for the new proposed system.

Some potential fluids, such as ethane, propane, sulfur hexafluoride and HFC refrigerants, may be candidates of the phase-change component. But it is pointed out that most organic fluids, including isobutane, exhibit the behavior of low fluid stability, high flammability, and corrosiveness near or at supercritical pressure operation [24]. Thus, when a working fluid is used in the new proposed system, the behavior near or at supercritical pressure should be accounted for. The high flammability of single fluid can be obviated by using blended fluid (mixture) in the proposed system, as done in the refrigeration and air-conditioning system at present [21]. In our preliminary calculation, the maximum supercritical pressure of fluid is about three times higher than the critical pressure, which is far from the critical state. The transport properties at this kind of supercritical state are different from that at sub-critical state or near supercritical pressure operation [10]. Further careful attention should be paid to the transport properties, such as small viscosity at supercritical state for small pressure loss in the heat exchanger.

Stirling engine cycle, using single multi-phase working fluids as the working fluids, may be named as the regenerative supercritical power cycle.

7. Conclusions

Using a supercritical heat-recovery and heating process, the composite working fluids, including two-phase two-component fluids and single multi-phase fluids, can be effectively used in Stirling engines. The principles are demonstrated compared to common gaseous Stirling engines.

In the Stirling engine with single multi-phase fluid, a suitable working fluid, the appropriate condensing temperature (pressure) of phase-change component, and consequently the low latent

heat, could be chosen to promote the thermal efficiency of Stirling engine to a high value. The minimum pressure, one of the important cyclic parameters of Stirling engine should be decided with no freedom by a trial and error calculation process of cyclic parameters. This calculation process can be programmed.

The criteria for the choice of working fluid are also formulated in this paper. Preliminary thermodynamic calculation, using sulfur hexafluoride (SF_6) as the working fluid, is carried out to show the cyclic features, such as the thermal efficiency and the optimum condensing temperature (pressure).

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